

# Modifications to Marshall's Annular Seal Test (MAST) Rig and Facility for Improved Rotordynamic Coefficient Testing of Annular Seals and Fluid Film Bearings

The limits of rotordynamic stability continue to be pushed by the high power densities and rotational speeds of modern rocket engine turbomachinery. Destabilizing forces increase dramatically with rotor speed. Rotordynamic stability is lost when these destabilizing forces overwhelm the stabilizing forces. The vibration from the unstable rotor grows until it is limited by some nonlinearity. For example, a rolling element bearing with a stiffness characteristic that increases with deflection may limit the vibration amplitude. The loads and deflections resulting from this limit cycle vibration (LCV) can lead to bearing and seal damage which promotes ever increasing levels of subsynchronous vibration. Engineers combat LCV by introducing rotordynamic elements that generate increased stabilizing forces and reduced destabilizing forces. For example, replacing a labyrinth seal with a damping seal results in substantial increases in the damping and stiffness rotordynamic coefficients. Adding a swirl brake to the damping seal greatly reduces the destabilizing cross-coupled forces generated by the damping seal for even further increases in the stabilizing capacity. Marshall's Annular Seal Test (MAST) rig is designed to experimentally measure the stabilizing capacity of new annular seal designs. The rig has been moved to a new facility and outfitted with a new slave bearing to allow increased test durations and to enable the testing of fluid film bearings. The purpose of this paper is to describe the new facility and the new bearing arrangement. Several novel seal and bearing designs will also be discussed.

# **MODIFICATIONS TO MARSHALL'S ANNULAR SEAL TEST (MAST) RIG AND FACILITY FOR IMPROVED ROTORDYNAMIC COEFFICIENT TESTING OF ANNULAR SEALS AND FLUID FILM BEARINGS**

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## **ABSTRACT**

The limits of rotordynamic stability continue to be pushed by the high power densities and rotational speeds of modern rocket engine turbomachinery. Destabilizing forces increase dramatically with rotor speed. Rotordynamic stability is lost when these destabilizing forces overwhelm the stabilizing forces. The vibration from the unstable rotor grows until it is limited by a stiffening bearing characteristic, or some other nonlinearity. The loads and deflections resulting from this limit cycle vibration (LCV) can lead to bearing and seal damage which promotes ever increasing levels of subsynchronous vibration. Engineers combat LCV by introducing rotordynamic elements that generate increased stabilizing forces and reduced destabilizing forces. For example, replacing a labyrinth seal with a damping seal results in substantial increases in damping and direct stiffness. Adding a swirl brake to the damping seal greatly reduces the destabilizing cross-coupled stiffness forces generated by the damping seal for even further increases in stabilizing capacity. Marshall's Annular Seal Test (MAST) rig is designed to experimentally measure the rotordynamic characteristics of new annular seal designs. The rig has been moved to a new facility and outfitted with a new slave bearing to allow increased test durations and to enable the testing of fluid film bearings. The purpose of this paper is to describe the new facility and the new bearing arrangement.

## **MOTIVATION FOR MODIFICATIONS**

Marshall's Annular Seal Test (MAST) rig was designed and developed to measure the rotordynamic coefficients associated with turbulent-flow annular seals. Previous investigations into annular seal rotordynamic performance using this rig have yielded measurements on both smooth and round-hole pattern seals, seals with different radial clearances, seals with a convergent radial clearance, as well as an understanding of influence of swirl brakes on the seal's capacity to stabilize (References [1-5]). For future experimental investigations, however, several modifications have been made to the rig's supporting infrastructure as well as to the rig itself. These modifications were motivated by several factors including high recurring test costs, inability to perform multiple tests within a reasonable time frame, and the limited capability of the test rig to accept other forms of fluid film bearings. The original infrastructure proved to have high recurring test costs as well as requiring significant manpower to operate. In terms of performing multiple tests, the original facility required considerable effort to prepare for a second test within the same day. This limitation proved costly for several reasons including instrumentation issues, data recording issues, electrodynamic shaker problems as well as others. Furthermore, review of the measured data has prompted questions that could be evaluated with additional testing. The inability to perform tests in a timely manner has prevented some additional diagnostic tests that could have been used to validate certain data sets. Finally, the test rig was designed to evaluate high pressure-drop annular seals to gain insight into their rotordynamic characteristics. Because annular seals are generally used to separate high pressure volumes from low pressure volumes, the test rig was designed to accommodate the large static load associated with the pressure difference across the annular seal. Additionally, the upper slave bearing that supports the rig's shaft also is dependent on the high pressure region upstream of the test article for its working fluid thereby creating a coupling between the slave bearing and the test article. Thus, the rig has been modified to incorporate an externally-supplied fluid film bearing (as opposed to being supplied internally) as well as an externally-supplied upper slave bearing for rotor support.

## TEST RIG MODIFICATIONS

### Test Rig Capabilities

Motivated by vibration issues on the turbomachinery from the Space Shuttle Main Engine (SSME), the turbomachinery from the SSME was used as an approximate basis for the design of this test rig. Both the rig's shaft/seal arrangement and pressure levels are consistent with the turbomachinery found on the SSME. The working fluid for the rocket engine (typically liquid oxygen or liquid hydrogen) would be the natural test medium, however, prohibitive costs and additional test complexity dictate that water be used as the test fluid. The density of water is relatively close to that of liquid oxygen but the viscosity differs significantly between the two. To achieve sufficiently turbulent flow in the test rig, seal clearances are generally increased beyond typical turbomachinery applications to yield adequate axial velocities. Further, high pressure differences across the inlet and exit of the annular seal on the order of 2000 pounds per square inch (psi) help to ensure the flow is sufficiently turbulent. Analytical models anchored with previous experimental data are relied upon to predict coefficients for both fuel and oxidizer turbomachinery configured with annular seals.

The test rig consists of a vertical shaft rotating up to 18,000 revolutions per minute (rpm) supported by fluid film bearings at the top and bottom of the shaft. Although there is great variability in the rotational speeds found in various rocket-engine turbomachinery, the rig's rotational speed is representative. One of the most important design features of this test rig is its ability to replicate a turbopump environment by imparting a significant fluid circumferential velocity just upstream of the test article. Labyrinth and annular seals can generate significant destabilizing forces that promote rotordynamic instability in turbomachinery due to high levels of circumferential fluid velocity entering these types of high-pressure seals. These forces have resulted in destructive vibration that damaged engines and significantly delayed development programs. Thus, it is imperative that a test rig is designed to provide a quantifiable metric into a given seal's ability to stabilize.

An effort, described within this work, is underway to upgrade the test rig to enable experimental evaluation of fluid film bearings as well as the annular seals. Also, modifications to the rig's infrastructure are aimed at reduced recurring costs as well as the ability to perform more extensive testing.

### Rotordynamic Coefficient Measurements

The fundamental philosophy that guided the design of the test rig is to effectively isolate the test article from the remainder of the test rig. The goal of isolation is to minimize external forces that can compromise the quality of the measured force which, if corrupted, would have an impact on resulting coefficient estimates. Stated differently, higher quality experimental estimates result from force measurements that are due to the seal's fluid film reaction force alone. Thus, external forces such as friction and tare procedures that can corrupt the measurement are minimized and/or eliminated. Likewise, four eddy-current proximity probes are mounted in 90 degree increments in the axial center of the test article to provide the highest quality measured displacements. These probes allow direct measurement of the relative shaft/seal displacement. Past experience with the probes has been excellent in the sense that probes separated by 180 degrees possess excellent agreement. Isolation of the test article in conjunction with direct displacement measurements lead to a relatively straightforward methodology to obtain coefficient estimates. This methodology was discussed in detail in reference [1].

An electrodynamic shaker is used to apply a radial load to the test article through a lever system with a mechanical advantage of six. Two piezoelectric force transducers are located between the output side of the lever system and the test article providing a measurement of the loads imparted to the test article. These transducers are mounted inline on rods with spherical bearings on both rod ends to relieve any residual moment. A band-limited white noise excitation is used as an input force to the test article ranging from 20 to 200 cycles per second. The coefficient estimation methodology naturally requires the force due to the inertia of the test article (and its associated mounting hardware) be removed from the measured force. To that end, the radial displacement of test article is measured with respect to the rig's housing. To

ensure the displacement (and, ultimately, acceleration) is measured with respect to an inertial frame of reference, accelerometers are used to measure any acceleration of the rig's housing.

## Hydrostatic Bearing Test Capability

Historically, the MAST rig has been used to measure the rotordynamic coefficients of various configurations of annular seals. Improving the rotordynamic performance of annular seals has enhanced the stability margins for many turbopumps that have traditionally been supported by rolling element bearings. Many new turbopump designs employ hydrostatic bearings, however, due to their stiffness characteristics, good stability properties, and long life. In its original configuration, the MAST rig cannot support the experimental evaluation of hydrostatic bearings for several reasons. The primary reason is the manner in which high pressure water supplies both the test article and upper slave bearing from a common plenum. With the upper slave bearing and the test article being axially-fed annular seals, the introduction of an internally-fed hydrostatic bearing requires significant change to the rig's flowpath (see Figure 1).

To enable a hydrostatic bearing to function properly within the rig, pressure within the common plenum must be reduced to provide a lower sump pressure on the plenum side of the bearing while maintaining the low pressure at the opposite discharge plane of the bearing. Since high pressure water will be externally-supplied to the hydrostatic bearing's manifold (which supplies the bearing's orifices), the volume that previously served as the common plenum will, in the new configuration, become a common drain. If not redesigned, the upper slave bearing could not function properly because it requires a high pressure source of water from the common plenum. Thus, a new upper slave bearing (see Figure 2) has been designed and fabricated which is comprised of adjacent damping seals separated by a central galley. The galley is externally-supplied with high-pressure water which flows through both seals but in opposite flow directions that, ultimately, discharge into upper and lower drains (the lower drain being the common plenum). The new upper slave bearing allows the evaluation of hydrostatic bearings for the first time but maintains the capability to continue testing variations of annular seals.

Testing annular seals in the new configuration requires high-pressure water within the common plenum to provide the necessary flow through the annular seal test article. However, this flow configuration also results in a high-pressure sump for the discharge of the lower-half of the new configuration of the upper slave bearing. This issue is alleviated by the fact that the upper-half of the new slave bearing (that discharges to atmospheric pressure) has the capacity to support the rotor by itself. The original slave bearing and housing have been retained in case there are some unforeseen consequences associated with operation of the new slave bearing.

## New Thrust Hydrostatic Bearing

As mentioned previously, the overriding design philosophy for the test rig was to minimize the external forces that act on the test article (and associated hardware) so that the mechanical impedance of the seal can be accurately characterized. This design philosophy can be seen clearly in mounting hardware that incorporates the test article. This mounting hardware, referred to as the seal carrier, has circular, flat surfaces on both its upper and lower faces with the test article situated within this assembly (see Figure 3). Upon introduction of the seal carrier into the test rig, these flat surfaces interface upper and lower thrust hydrostatic bearings. The purpose of the hydrostatic bearings is twofold: to react the substantial static load generated by the 2000 psi pressure differential across the test article and to provide a means for the test article to translate (due to excitation from an electrodynamic shaker) without significant resistance. Without significant resistance to translation, the applied forces, measured via piezoelectric force transducers, are reacted by the mechanical impedance of the test article but also include the seal carrier's inertial force. The measured forces are corrected for the seal carrier's inertial force as described in reference [1].

Over the course of the test rig's operational life, the lower hydrostatic bearing (whose job is to react the pressure-generated static load) has failed to lift the seal carrier to allow translation on several occasions. In each case, the lower hydrostatic bearing was found to be at fault. The issue centers on the

construction of the lower bearing having a brass bearing surface brazed to a solid stainless steel piece. The bearing failed due to leakage of water through the brass-stainless steel interface which, as a consequence, reduced the flow of water through the bearing's orifices. Without an adequate flow through the hydrostatic bearing, the bearing could not react the static load acting on the seal. To correct the situation, a replacement lower hydrostatic bearing was fabricated from a solid piece of bronze (see Figure 4). Subsequent testing of the new lower hydrostatic bearing in the test rig confirmed its operational viability.

## FACILITY MODIFICATIONS

### Original Facility Configuration

Since the initial attempts to experimentally evaluate annular seals for rocket-engine turbomachinery, the facility used to support the test rig consisted of a 250 horsepower steam turbine to power the rig's shaft and a 3000 gallon tank pressurized with gaseous nitrogen to provide water at sufficient pressure levels. Although these components provided excellent performance characteristics they also were found to be prohibitively expensive. For example, the steam turbine provided smooth, vibration-free power to rotate the shaft to speeds as high as 22,000 revolutions per minute (rpm). However, the steam required to power the turbine was generated by a boiler system that consumed significant quantities of fuel and the appropriate personnel to oversee the steam generation. Similarly, the blow-down system provided smooth pressure at levels exceeding 2000 psi to the test rig. Unfortunately, the high-pressure water tank required a bank of nitrogen bottles pressurized at 5000 psi which proved costly as well as time consuming to reconfigure for an additional test. Both of these systems had inherent recurring costs that became cost-prohibitive over time (see Figure 5).

### New Facility Configuration

In contrast with the previous facility, the new infrastructure is intended to minimize the recurring test costs to allow for more extensive testing. Since the most significant recurring costs involved powering the test rig's shaft and achieving the desired pressure levels and flowrates, these systems have been replaced (see Figure 6). Shaft power is now provided by 300 HP variable speed electric motor with a maximum speed of 3600 RPM. A low speed coupling connects the motor to a speed increasing transmission with a gear ratio of 1 to 5.07. From there the power is turned from horizontal to the vertical direction via a bevel gear transmission. A shrouded high speed coupling connects the bevel gear transmission to the test rig shaft. The costs associated with operating the new electrical drive system are intended to be lower than the original steam system. There is concern, however, that bearing and gear noise could possibly corrupt the measured data resulting in poor coherence between the applied seal forces and the resulting displacements. To mitigate this risk, the motor and transmissions are mounted to the laboratory's cement floor. Thus, structural-born noise will have to propagate through the floor or the high speed coupling to reach the rig to influence the measurements. Coherence data is not yet available to determine the success of this layout. If data is lost in specific frequency ranges, the excitation frequency range can be tailored to achieve coherent data for impedance calculations at higher frequencies.

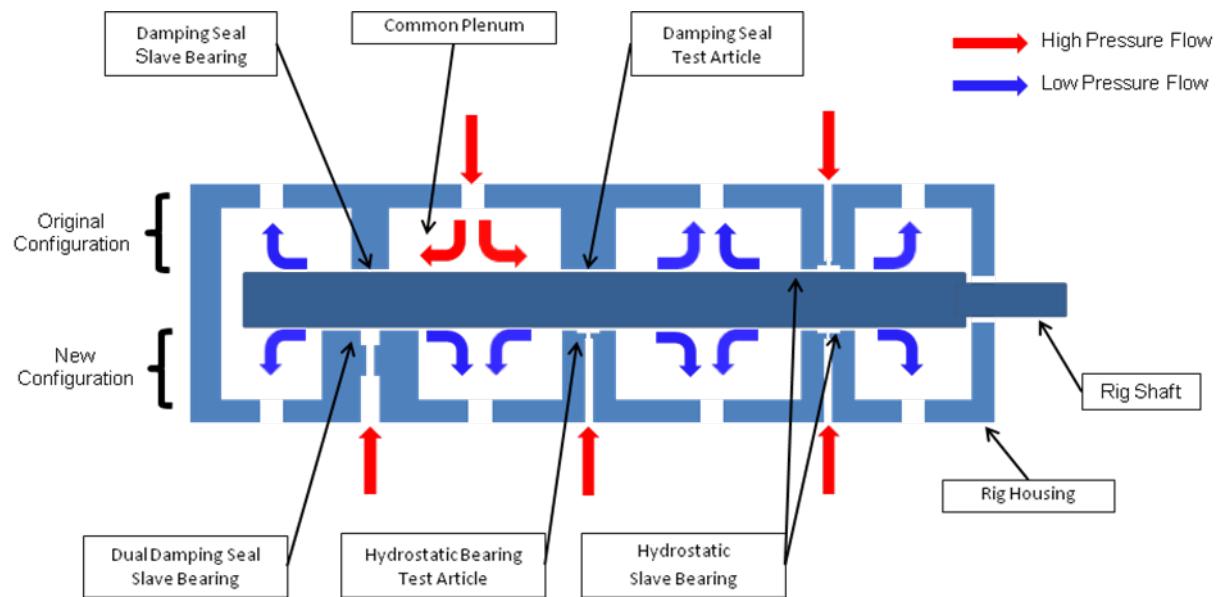
To achieve the proper pressure level and flowrate to the test rig, a collection of ten positive displacement pumps are used to provide over two hundred gallons per minute (gpm) to the test rig. This arrangement, on paper, yields smaller recurring costs, but introduces pressure pulsations from the pumps have the potential to disrupt the measured dynamic data. To mitigate this risk, each pump was outfitted with a soft inlet line that can collapse and expand slightly to reduce the cavitation induced pressure perturbations generated by the pump. The discharge line of each pump was also outfitted with a nitrogen-charged accumulator to dampen pressure pulsations. In an effort to be thorough, pressure pulsations were measured at the inlet manifold to the test rig with a Kistler piezoelectric dynamic pressure transducer that yielded a maximum pulsation of only 0.4 psi. Although encouraging, coherence data has yet to be obtained to verify there are no deleterious effects on the quality of the resultant coefficient estimates.

## SUMMARY

The MAST rig and its supporting infrastructure have undergone significant modifications with the goals of enhancing the rig to accept an additional class of fluid film bearings, increasing test time, and reducing recurring test costs. In order to facilitate the rig's ability to evaluate fluid film bearings, its upper slave bearing and major flow paths within the rig required modification. In its updated configuration, the rig will be able to evaluate fluid film bearings that require an external source of working fluid such as a hydrostatic bearing. Next, experimental evaluations of bearings and seals will benefit from additional test time by providing checks on measured data that appears unusual. This additional time should prove helpful with more common situations such as problems found in the measured data and instrumentation, as well as other issues. Finally, to reduce recurring costs, the transition to a pump-fed rig and the use of a combination of electric motor and transmission should prove beneficial. The original facility used to support the MAST rig provided smooth power to rotate the rig's shaft and the blow-down system supplied the rig with a constant pressure level without any detectable pulsations. This performance, however, came at a cost. The required personnel to conduct a test and the turn-around time to recycle the facility for additional testing was prohibitive. The updated facility utilizes pumps to generate the required pressure and flowrate and can be operated with fewer personnel. Likewise, the replacement of the steam-driven rig shaft with an electric motor coupled to a pair of transmissions reduces the operational overhead by requiring fewer personnel and lower operational costs.

## REFERENCES

1. Darden, J. M., Earhart, E. M., Flowers, G. T., **Experimental Rotordynamic Characterization of Annular Seals: Facility and Methodology**, ASME Journal of engineering for Gas Turbines and Power, Vol. 121, April 1999, pp. 349-354.
2. Darden, J. M., Earhart, E. M., Flowers, G. T., **Comparison of the Dynamic Characteristics of Smooth annular Seals and Damping Seals**, ASME Journal of Engineering for Gas Turbines and Power, Vol. 123, October 2001a, pp. 857-863.
3. Darden, J. M., Earhart, E. M., Flowers, G. T, **Influence of Seal Geometry on the Rotordynamic Characteristics of a Round-Hole Pattern Damping Seal**, Proceedings of the 2001 ASME Vibration conference. DET2001/VIB-21635.
4. Darden, J. M., Earhart, E. M., **The Effect of a Swirl Brake on the Dynamic Characteristics of Smooth Annular Seals and Damping Seals**, JANNAF 6th Modeling and Simulation Subcommittee / 4th Liquid Propulsion Subcommittee / 3rd Spacecraft Propulsion Subcommittee Joint Meeting, December, 2008
5. Von Pragau, G. L., **Damping Seals for Turbomachinery**, NASA Technical Paper 1987, 1982.



**Figure 1 Comparison of MAST rig's original and updated configuration**



**Figure 2 Dual damping seal upper slave bearing**

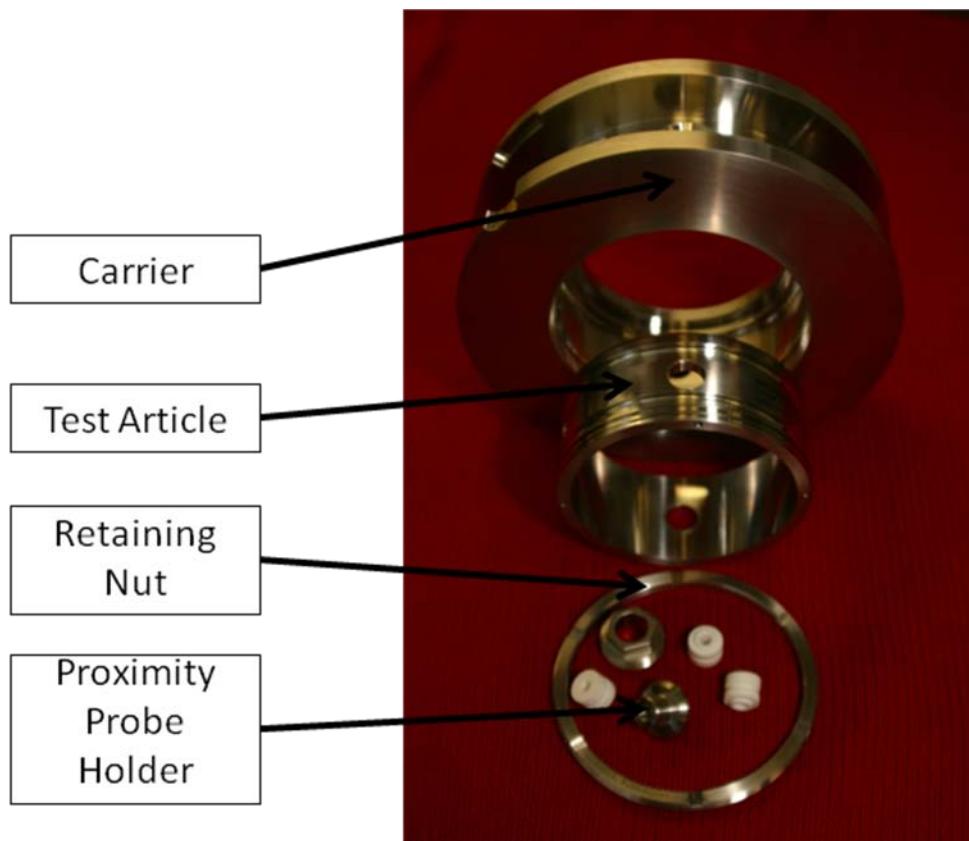


Figure 3 Seal carrier assembly



Figure 4 Lower hydrostatic thrust bearing

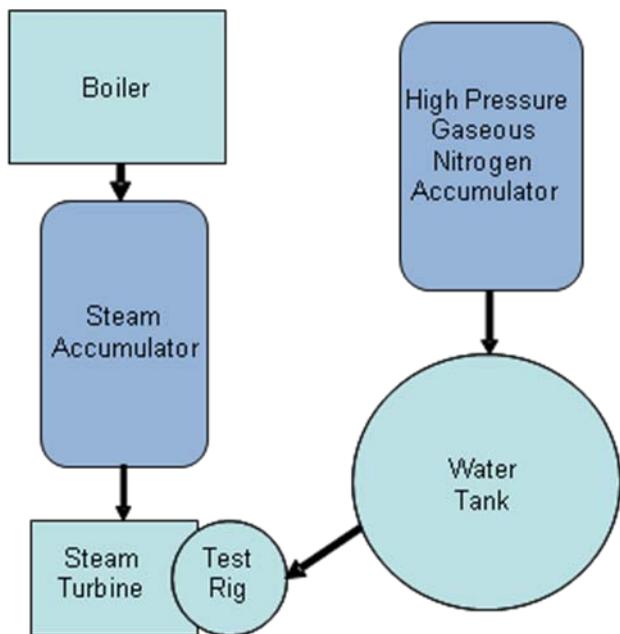


Figure 5 Original facility layout

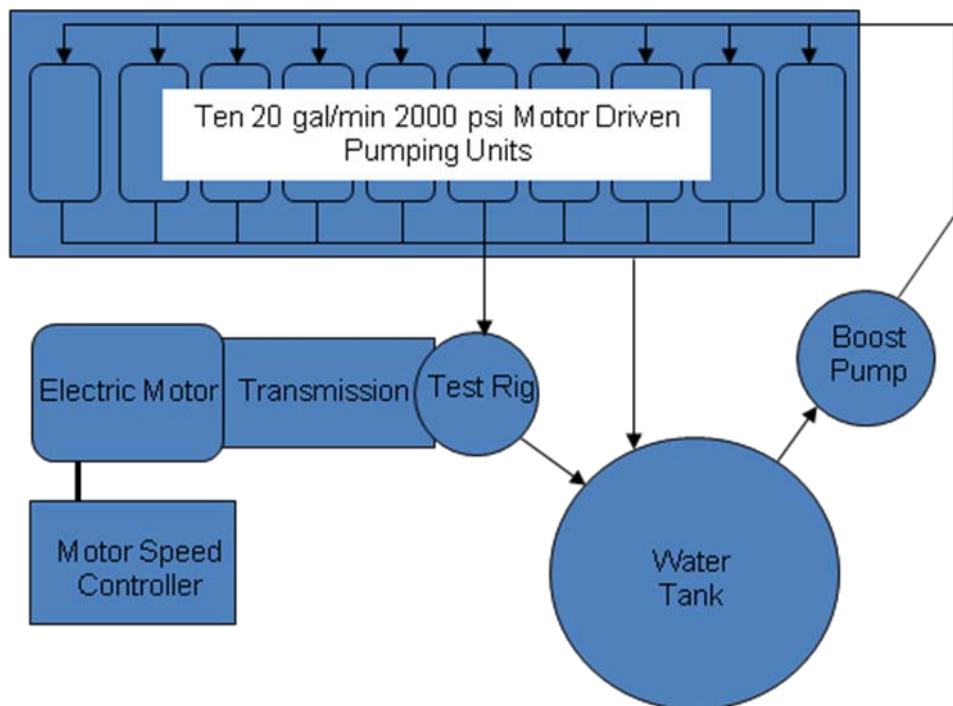


Figure 6 Updated facility layout